

A Compact Roller-Gear Pitch-Yaw Joint Module: Design and Control Issues

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Abstract

Robotic systems have been proposed as a means of accomplishing assembly and maintenance tasks in space. Desirable characteristics of these systems include compact size, low mass, high load capacity, and programmable compliance to improve assembly performance. In addition, the mechanical system must transmit power in such a way as to allow high performance control of the system. Efficiency, linearity, low backlash, low torque ripple and low friction are all desirable characteristics. This work presents a pitch-yaw joint module designed and built to address these issues. Its effectiveness as a two degree-of-freedom manipulator using natural admittance control, a method of force control, is demonstrated.

1 Introduction

The use of robotic manipulators is attractive in space operations, whether as autonomous systems, or as tele-operated devices. Possible uses include assembly, repair and servicing, inspection, retrieval and exploration. Much effort has been devoted to the development of controls to accomplish suitable tasks. Somewhat less has been expended for improving the mechanical components in robotic systems. The interaction of the controls and the mechanical components' characteristics (i.e.: stiffness, backlash, friction) is often a serious issue.

The major task in any control scheme is enforcing the desired behavior of the controlled system in the presence of disturbances: gravity, end effector forces, and collisions with obstacles; and internal dynamics: friction, mechanical stiffness, transmission dynamics, actuator dynamics, sensor noise, and time delay. Position control attempts to force a specific end-effector trajectory. Force control tries to enforce a specified end-effector contact force. However, classical approaches to force control have had severe limitations and problems with instabilities [1].

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An alternate approach to direct or explicit force control is to control not a vector of positions or forces, but the relationship between those variables using a programmable compliance or impedance [2, 3]. The controller specifies the manipulator's response to environmental disturbances in some desired way, usually emulating programmable springs and dampers. In impedance control, the stiffness and damping parameters can be adjusted to the task at hand: high stiffness for precise positioning, and low stiffness to accommodate interaction forces. However, stability of the system when interacting with its environment is still a problem [4, 5, 6, 7, 8] If the controlled system is to remain stable when coupled to an arbitrary passive environment (one that can be modeled using only passive elements), it must itself present a passive mechanical admittance to the environment at the end effector [7]. This imposes restrictions on how the intrinsic endpoint dynamics of a manipulator may be modified by the control. In particular, reducing the apparent (i.e. desired) inertia can result in a system that is not passive which will be unstable when coupled to some passive environment.

One remedy is to accept the natural manipulator inertia and make no attempt to change it. However, this is often disappointing on machines which have large amounts of friction and other non-linearities such as gearing backlash. Friction limits sensitivity to small end effector forces and backlash makes stable control more difficult. Robots can be designed with lower friction and direct drive actuators to reduce transmission dynamics, but payload capacities are reduced, and the motors become larger and heavier. This raises several questions. Is it possible to compensate for friction and other effects without violating passivity? If systems must contain high friction components to meet payload, weight, and size requirements, can they be designed to make the compensation task easier? What approach can be used to get the maximum possible performance, given a particular manipulator system?

Natural admittance control (NAC) [9, 10] addresses these issues. Using this approach, it is not only possible to improve performance while maintaining stability, but also to identify mechanical design guidelines that improve ultimate performance. Also, it suggests that the endpoint admittance is a good performance measure.

In this paper, we review the design of a 2-degree-of-freedom (2-DOF) pitch-yaw roller-gear joint module which has been developed to exhibit low friction and no backlash. The application of natural admittance control to the transmission will also be examined. Resulting system performance will be presented, along with recommendations for other designs.

2 Two Degree of Freedom Arm

The pitch-yaw module is part of a 2-DOF manipulator and was designed and built by NASTEC under contract to NASA Lewis Research Center [11]. It is similar in specifications to the large joint of the Micro-Gravity Manipulator and the Laboratory Telerobotic Manipulator built by Oak Ridge National Laboratories for NASA-Lewis and NASA-Langley, respectively [12, 13], but uses roller-gears for torque transmission. Transmissions of this type have been shown to have some of the smoothness of pure traction drive

transmissions with higher torque capacity at a given roller loading due to the gear teeth [14]. The module input is driven by a pair of gear motor drives having a 90:1 ratio in the gear heads. A link is mounted to the pitch-yaw module output, creating a 2-DOF manipulator arm, capable of moving payloads through a spherical-surface workspace. Overall, the joint is capable of lifting 220 Newtons (50lbs) at a moment arm of 0.84m (33in), the length of the link.

2.1 Pitch-Yaw Module Design

The pitch-yaw module provides two degrees of freedom in a compact space. Figure 1 shows a cross-section of the components in the module. In this drawing, the roller and

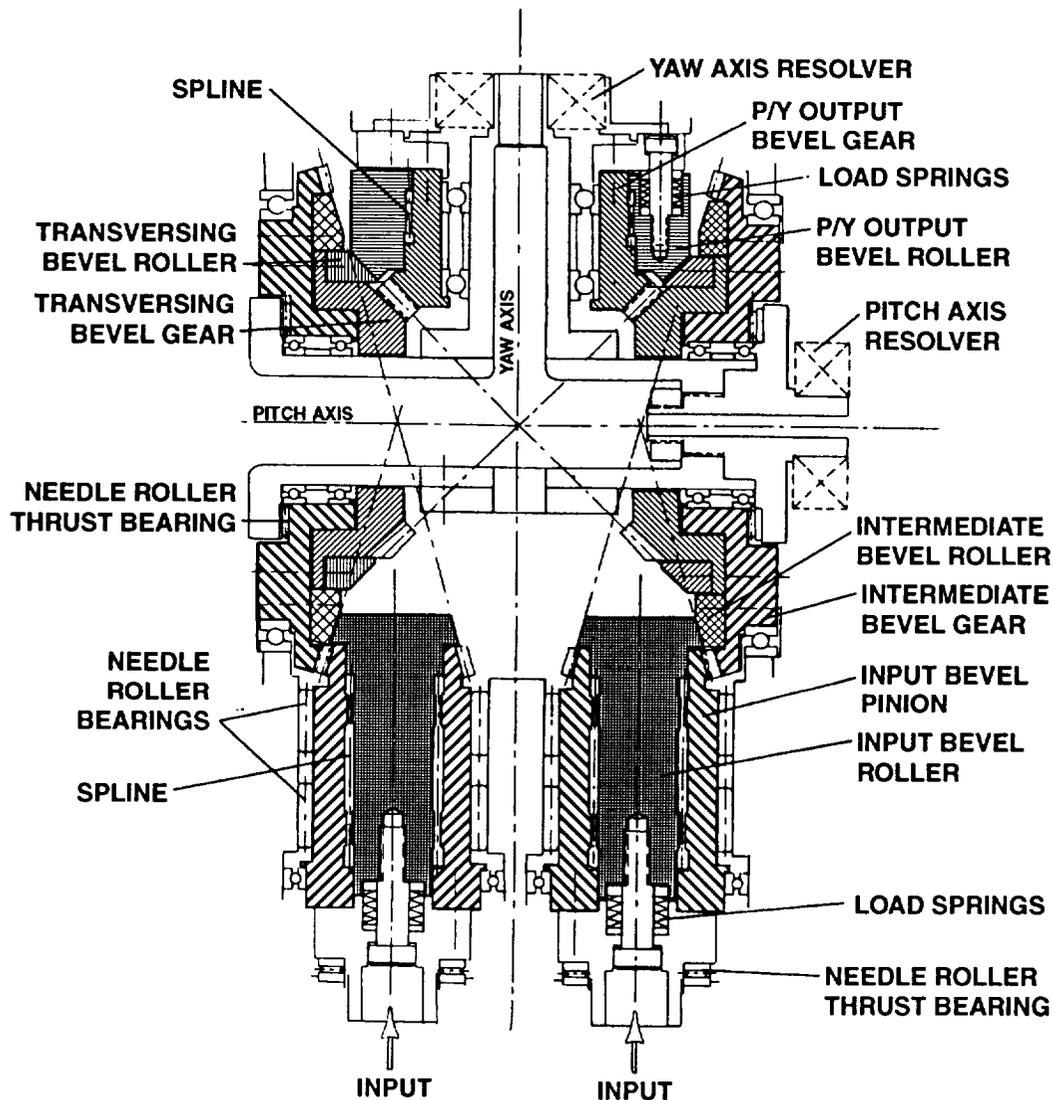


Figure 1: Drawing of pitch yaw module

gear elements are cross-hatched, and much of the housing detail is omitted for simplicity. The module is a differential, having two parallel bidirectional inputs and a crossed-axis output which moves in pitch and yaw. Referring to the figure, the input bevel pinion gears drive the intermediate bevel gears. The input bevel rollers are splined to the pinion gears, and are spring-loaded axially against the intermediate bevel rollers. On each side of the module, the intermediate gear and roller are fixed to each other as well as to the transversing bevel gears and rollers. These intermediate/transversing assemblies are not, however, fixed to each other, but rotate individually about the pitch axis. Both transversing bevel gears drive the pitch-yaw output gear. The output roller is spring-loaded against the transversing bevel rollers and moves axially on the bevel gear via a spline. Rotation of the two inputs in the same direction at the same speed results in a pure yaw output; rotation in opposite directions at the same speed results in a pure pitch motion. All other combinations of input speed and rotation result in motions about both the pitch and yaw axes. The speed ratio across the module to either a pure pitch or pure yaw output is 3.4283.

Figure 2 shows a photo of the module partially assembled. In this orientation, the two

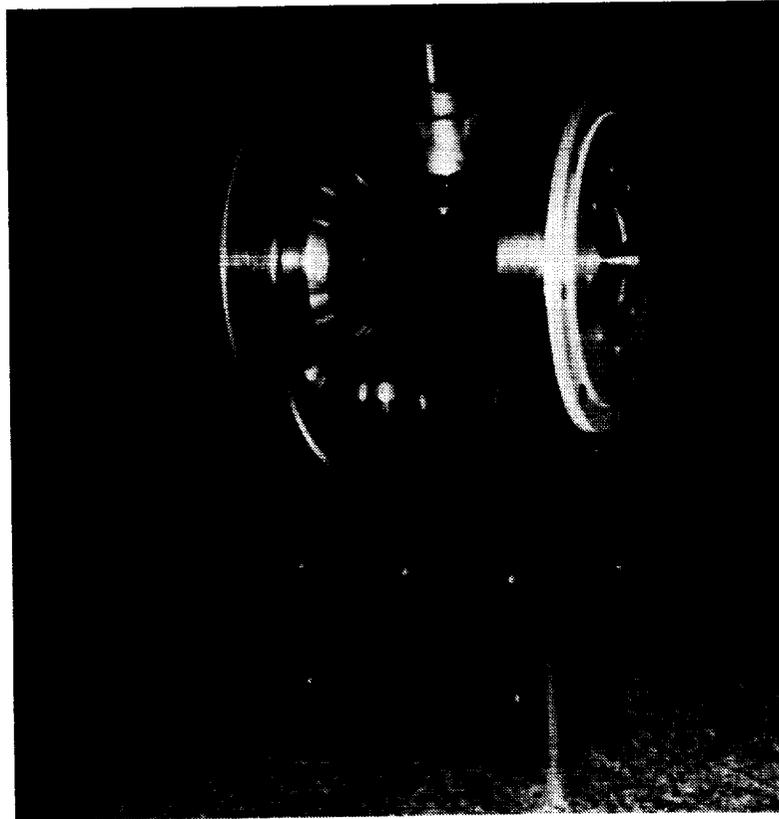


Figure 2: Photo of pitch yaw module

input members are hidden at the bottom with axes vertical. The pitch-yaw output bevel-roller is on top, with the yaw axis vertical and the pitch axis horizontal. A bracket (not shown) is bolted to the output member, to which the link is attached. When integrated in the manipulator arm, the module's pitch axis, yaw axis, and the centerline of the

output link are mutually perpendicular. The input members are each driven by a 90:1 ratio gearbox at a maximum input speed of 2.3radians/sec (22rpm). Maximum input torques are 27.1 N-m (240 in-lbf). Maximum output torque is 186 N-m (1650 in-lbf) on each axis.

All gears are 20° pressure angle Zerol bevel gears of case carburized AISI 9310 steel. Gear data are as follows:

	Number of teeth	Diametrical pitch	Pitch dia., in.
Input Bevel Pinion (2)	28	16	1.750
Intermediate Bevel Gear (2)	96	16	6.000
Transversing Bevel Gear (2)	39	12	3.250
P/Y Output Bevel Gear	39	12	3.250

Rollers are also made of case carburized AISI 9310 steel with slightly crowned cones to produce elliptical contact areas. Rollers are loaded so that they can transmit 20 percent of rated torque at a traction coefficient of 0.06 with traction grease. This requires a normal force of 4900N (1100 lbf) at each input bevel roller/intermediate bevel roller contact (axial spring force of 1370N (308 lbf)), and a normal force of 6620N (1490 lbf) at each transversing bevel roller/output bevel roller contact (axial spring force per contact of 9360N (2100 lbf)). Other critical features of the roller-gear design include the following: precision setup of each bevel gear pair with fitted spacers for minimum backlash; precision control of roller-gear pair axis concentricity; and high capacity needle thrust and radial bearings for reacting input roller forces. Complete details are given in [11].

The value of having rollers in the drive train is twofold. First, they remove backlash from the system. At startup, when the gear teeth are not fully engaged, rollers transmit torque. Although the theoretical speed ratio of each roller pair is identical to that of its paralleled gear pair, rollers undergo a small loss in motion known as “creep” when transmitting torque. This allows the gear pair to “catch up”, and begin transmitting torque shortly after motion ensues. Second, they attenuate gear cogging and ripple in the output motion. Rollers accomplish this by reacting the radial component of gear tooth contact forces so that the teeth transmit purely tangential forces.

An alternative design which uses only rollers and transmits all torque through traction contacts will produce smoother motion that is completely free of gear induced ripple. A pure roller design, however, requires much higher magnitudes of normal forces at the roller contacts. This necessitates incorporation of a cam-actuated variable load system to maintain reasonable part load efficiencies, and larger bearings and heavier housings to minimize deflections. Also, high preloads result in high inherent friction. In comparison, the large, all-roller pitch-yaw joint, with a dry-lubed traction coefficient of 0.1 [12] had roller normal loads 2.5 to 3 times greater than those in the present roller-gear drive module. If the roller drive had been designed with a coefficient of 0.06, as was the roller-gear drive, the loads would have been even higher. Further, the present spring-loaded, constant roller-preload design represents a simplification over the variable preload mechanism.

2.2 Drive System and Instrumentation

The system is well-instrumented, making it useful for analyzing its characteristics and developing sophisticated control strategies. Each gear motor has a tachometer (as well as a brake) coupled directly to the motor shaft. The roller-gear inputs are driven through a torque meter, and a brushless resolver for measurement of input angle is built into each input drive train. As shown on figure 1, brushless resolvers are incorporated in both the pitch and yaw axes for measurement of output angle. For measurement of end-effector force, a load cell was mounted at the end of the link, although this signal was not used for control.

The module is controlled from a PC with custom made electronics for either preprogrammed tasks or operation with a joystick. A data processing system stores values from all system variables at 5 millisecond intervals for either digital printout or direct plotting of motor speeds, input angular positions, input torques, output pitch and yaw positions, output pitch and yaw velocities, and endpoint force parallel to the yaw axis.

2.3 Pitch-Yaw Module Tests

Static stiffness tests were conducted by fixing the inputs and measuring deflections of the output link in the pitch direction only under various dead-weight loads. The input pinion couplings were locked, and a .0001 in/div dial indicator was used to measure output deflections at a distance from the axis centerline of 0.84 m (33 in). The link and loading system represented a tare torque of 28.7 N-m (254 in-lbf) at the initially "zero" deflection point. Load torques of 9.29, 18.8, 37.2, and 58.4 N-m (82.2, 166, 329, and 517 in-lbf) were applied. The resulting deflection data were fairly linear, producing a least-squares fit for the slope (i.e.: stiffness) of 15,100 N-m/rad (134,000 in-lbf/rad).

By comparison, at full rated input torque, the theoretical stiffness of the gears was calculated to be 565,000 N-m/rad (5,000,000 in-lbf/rad). Since the rollers act in parallel with the gears, and are expected to have approximately the same order of magnitude stiffness as the gears [15], the stiffness of the roller-gear train should be higher than the gears alone. Note that this is for the rollers and gears alone. The large discrepancy between measured and theoretical is due to the fact that the theoretical value does not include the compliance of shafting, bearings, structure, and connections. It has been shown that the soft elements in similar systems are the structure and bearings [16]. The complex interaction of these components is beyond the scope of this paper. Backlash was also evaluated with the stiffness setup. No backlash was measured at the output pitch-yaw member with the inputs locked.

Efficiency of the pitch-yaw module was measured by summing the input torques from each input while lifting weights through a 30° angle with the link near horizontal. The loading system made it impossible to apply a torque about the yaw axis without also loading the pitch axis. Therefore for simplicity, tests were limited to pitch moves. The module was operated at four load levels: 0, 24.4, 81.9, and 137 N-m (0, 216, 725, 1210 in-lbf). Measurements were averaged during the constant speed portions of open-loop lifts, consisting of an acceleration, constant speed, and deceleration to stop. The total

lift move required less than one second; the duration of the constant speed portion was about 0.35 sec.

The total frictional loss at zero output load was determined to be .89 N-m (7.9 in-lbf). Efficiency data are plotted as a function of percent design load in figure 3. As

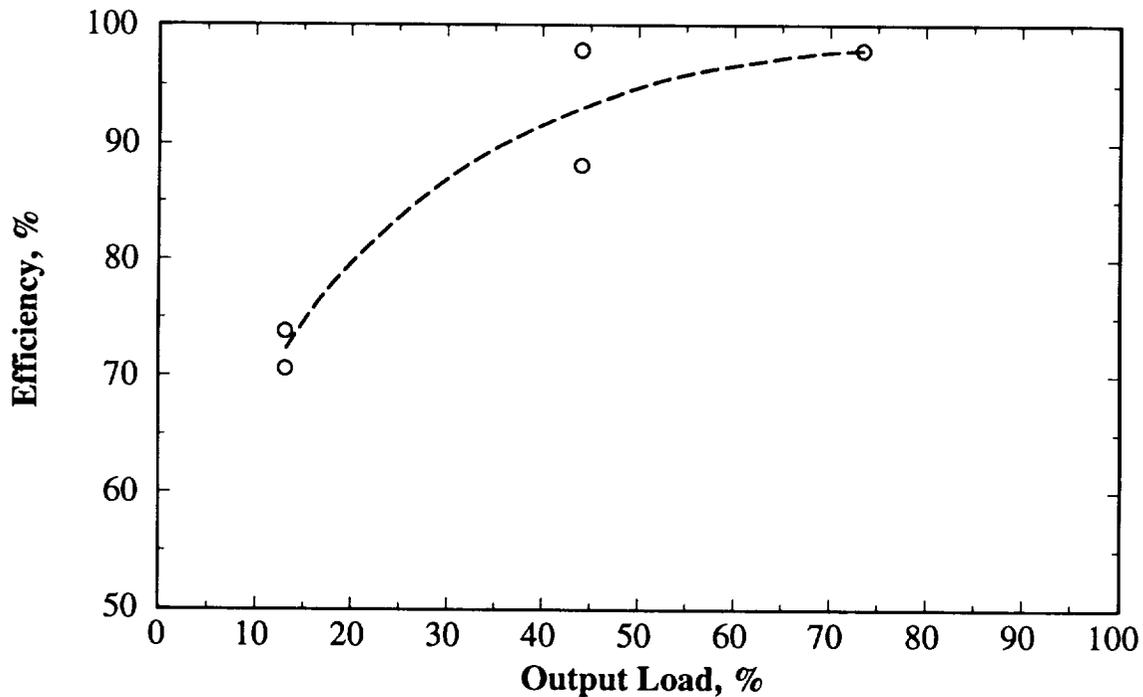


Figure 3: Pitch-Yaw Module Efficiency: Pitch Axis

would be expected for roller or gear systems, efficiency increases with increasing load. At approximately 50% load, measured values ranged from 88 to over 97% efficiency. The results indicate that pitch axis efficiencies of 98% or greater can be expected when the module is loaded at 80% or greater of full load rating.

2.4 Implications for Natural Admittance Control

This system is well-suited to NAC. Perhaps the most important feature is the location of the torque transducers between the gear-motors and the pitch-yaw module. Other experimenters have shown that closing torque or force loops around transmissions, not around links or whole manipulators, produces more robust force control [17, 18, 19]. In this configuration the pitch-yaw module and its link mass act as a filter for the environment, in effect becoming part of it. The filtering effect of the combination of link mass and transmission compliance makes the control more robust by limiting the apparent environmental stiffness. Further, endpoint dynamics are bandwidth limited by the physical dynamics between the sensor and the endpoint.

The pitch-yaw module's design qualities: smoothness, low-backlash, and moderate friction, provide a fairly linear, distortion-free filter. This is important since the location of the torque transducers makes it impossible to perform feedback compensation of any friction or other dynamics in the pitch-yaw module itself (or the link). Fortunately, most of the friction and backlash in the system resides in the gear-motor. This friction, located between the actuators and the torque transducers, *can* be compensated by the control using torque-feedback compensation.

An obvious question immediately comes to mind: why not put a sensor at the endpoint and compensate for all of the friction? The answer is fundamental to the nature of force control and the guiding principle behind NAC. Simply put, one can, but the limit on how far the dynamics at the end effector can be improved (i.e. friction reduced) is more restrictive. It is far more difficult to passively compensate for friction located beyond a major compliance in a mechanical chain. This has been analyzed elsewhere (see [20]), however to get a feel for why this might be so, consider the task of pulling a cinder block slowly along a floor using a bungee cord. As one begins to pull, the cord stretches, building up a force until it exceeds the maximum static friction between the block and the floor. The block then breaks free and begins to slide, possibly too fast requiring the actuator (person pulling the cord) to back off. The block then stops and the cycle repeats. The resulting motion is jerky instead of smooth.

Placing a force sensor at the distal end of the block and using it in a force feedback scheme is an exaggerated analogy to the task of force feedback around an entire manipulator. However, if one mounts the actuator firmly to the block, places the sensor, as before, on the other side, and connects the cord to the sensor and to a second, polished block, the task of making this second block move smoothly is much easier and much of the friction in the cinder block can also be masked. This is similar to the situation in the present mechanism. Most of the friction is close to the actuator. The torque sensor is next, and the output, the pitch-yaw module, is relatively low friction.

Another beneficial feature of this system for NAC is the selection of feedback sensors, which is larger than that in more standard manipulator designs. These sensors allow for the identification of important dynamic quantities used in the control, and also provide more flexibility for feedback. The tachometers, mounted directly on the motor shafts, are particularly important because the velocity loops closed around the motors include a minimum of unmodeled dynamics.

3 The controller

3.1 A simple model

In designing any controller, several goals must be met; a crucial one is stability. For systems that interact with the environment, stability must be assured not only for isolated operation, but also when the system is coupled to the environment. Guaranteeing coupled stability for an arbitrary environment is not possible; however, the class of environments presenting a passive impedance (i.e. one that could be constructed entirely

from passive elements) represents a useful class of target environments. By designing a controller that presents a passive endpoint admittance, coupled stability with any passive environment is assured. Passivity places limits on what is achievable in masking manipulator dynamics [6]. In particular, attempting to reduce the apparent mass of the manipulator is problematical at best and easily violates the passivity criterion, resulting in coupled instability for some passive environment.

Consider the simple, linear, 1 degree of freedom model of figure 4. This model represents a single mechanical link with one resonance. M_m is the actuator inertia reflected

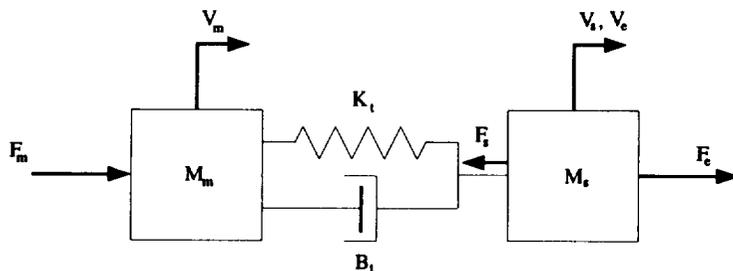


Figure 4: Simple two mass model

through the transmission, and M_s is the link inertia. F_m and v_m are the actuator effort and velocity, while F_e and v_e are the force and velocity at the end of the link. K_t and B_t represent transmission stiffness and damping. F_s represents the force in the spring-damper combination. This model can be used for one joint on any manipulator with rigid links and one dominant compliance in the transmission.

Feedback from an end-mounted effort sensor is often used to implement force control on an existing manipulator. This is the approach used in [10] where an implementation of NAC on a General Electric P-50 manipulator is presented in detail. The two-port admittance matrix was used to develop an appropriate control law using feedback from v_m and F_e . As shown in [10], the high frequency closed loop endpoint admittance approaches

$$Y_s = \frac{v_e}{F_e} = \frac{1}{M_s s} \quad (1)$$

This represent the maximum possible performance at low frequency as well since, increasing the low frequency admittance above this asymptote will violate passivity conditions.

Nevertheless, NAC can be successfully implemented on this type of architecture if this limit is accepted. However, if the sensor is moved closer to the motor, as in the manipulator in this work, measuring F_s instead, the control problem is not only simplified, but stability is easier to achieve, since the distal mass, M_s , and transmission compliance form a mechanical filter which effectively limits the environmental stiffness the control will encounter. Thus, passivity with respect to $\frac{v_e}{F_e}$ need only be maintained over a certain band of frequencies to achieve passivity with respect to $\frac{v_e}{F_s}$.

The following is a description of the natural admittance controller implemented on the pitch-yaw module manipulator. For convenience, all measurements were reflected

to the input pinions of the pitch-yaw module and converted to SI units, providing a consistent frame of reference for all dynamic quantities and eliminating the need for explicitly including the transmission ratios in the design.

3.2 Natural Admittance controller

A natural admittance controller operates by enforcing the manipulator's behavior to track a frictionless dynamic model of the manipulator, thus maximizing the end-point mechanical admittance. NAC also recognizes that there is a theoretical limit to what that maximum is. The limit (at low frequency, arguably the most important region for interaction tasks) is set by the intrinsic inertia of the manipulator as seen at the endpoint. In our case, most of the inertia in the system is the inertia of the gear-motors seen through the 90:1 transmission.

An inner proportional+integral velocity loop is given commands based on what the acceleration of the drive axis would be if there were no friction. The torque transducer signals are combined with gravity and friction estimates, as well as a virtual torque command derived from the desired endpoint stiffness and damping, to generate a net torque. The net torque is divided by the control's design inertia (normally the best estimate of the actual inertia) to generate a desired acceleration. This acceleration is integrated once for the velocity command, and a second time for the position (integral velocity) command.

If the velocity loop was perfect, then the manipulator would indeed appear frictionless from the torque transducer to the motor. Again, since the friction in the pitch-yaw module itself is outside the feedback loop, it is not compensated for in any way. However, making the velocity loop as good as possible motivated two key components of the control strategy: an observer to reduce tachometer signal noise, and a feed-forward friction model containing a dependence on the load applied. The friction model was used not only to reduce the burden on the velocity loop, but to improve the filtering capabilities of the observer. A more detailed presentation of the control design appears in [21], while a simple mechanical model offers an explanation of the load dependence of the friction appears in [22].

It is, in principle, possible to choose a control design inertia different from the actual inertia. Decreasing it would increase the low frequency admittance and help reduce impact transients. However, this will exceed the restrictions imposed by passivity requirements and lead to instability in contact with some stiff but passive environments. It is possible to increase the design inertia higher than the actual, but this will reduce the endpoint admittance at low frequency.

The endpoint's desired stiffness and damping can be controlled, however. This is done using a virtual reference frame (i.e. desired endpoint position and velocity) which is connected (mathematically) to the actual endpoint position and velocity through virtual, and programmable, springs and dampers. A correcting force is calculated based on the deviation of the desired and actual endpoint states and the desired stiffness and damping characteristics. This force is projected mathematically to the individual drive axes as torque commands and added to the other torques (sensed, gravity, and friction)

in the NAC algorithm. Thus, dragging the virtual frame around the workspace causes the manipulator to follow, but, when obstacles (or work surfaces) are encountered, the endpoint force is regulated by the desired stiffness and damping characteristics chosen by the operator.

3.3 Friction Measurements under Natural Admittance Control

With the virtual end-point stiffness and damping set to zero, the manipulator would ideally behave like a pure inertia. In reality, it comes quite close. The friction in the gear-motor drives is nearly completely masked, leaving only the much lower friction in the pitch-yaw roller-gear module.

Measurements were taken to determine the extent of the natural admittance controller's effect on the gearbox friction. A force was applied through the load cell to impart a slow, constant-velocity pure pitch motion. With NAC off, a force of 67N at the endpoint was required to overcome the friction of the drive trains and establish motion. Turning NAC on reduced the force required to about 2.5N. These figures translate into expected pinion torques (assuming lossless transmission through the differential) of 8.3N-m (NAC off) and 0.31N-m, (NAC on). The torque transducer signals were 6.5N-m and 0.07N-m for the two cases, implying frictional torques in the pitch-yaw differential of 1.8N-m and 0.24N-m.

The reduction in the required effort at the endpoint (and similarly the reduced torque measured at the pinion roller-gears) indicates that the apparent friction in the gear-motor drives was attenuated by more than an order of magnitude. Therefore the minimum force the manipulator can present to the environment under NAC is now limited mainly by the friction in the pitch-yaw differential itself. Since this friction is inherently low, the end-point force deadband is also low, and the system using NAC exhibits good sensitivity to end-point forces.

This result illustrates the advantage of placing the majority of the mechanical friction in a manipulator close to the actuator, between the actuator and the torque sensor. Friction beyond the torque sensor is not compensated for in the present implementation. One may well ask why not? Why not use the endpoint sensor in the natural admittance controller? One could, but doing so forfeits the robustness enhancing effects of a mechanical filter between the sensor and the environment.

3.4 Stiff Environment Contact Tests

To test the ability of our control to exhibit stable behavior when interacting with stiff environments, the endpoint was programmed for moderate stiffness and damping and commanded to collide with a rigidly mounted aluminum angle bracket at moderate speed. The load cell mounted on the end of the link measured the contact force. As shown in figure 5, the contact force reached a stable final value.

To explore the effects of incorrectly estimating the inertia, we repeated the experiment

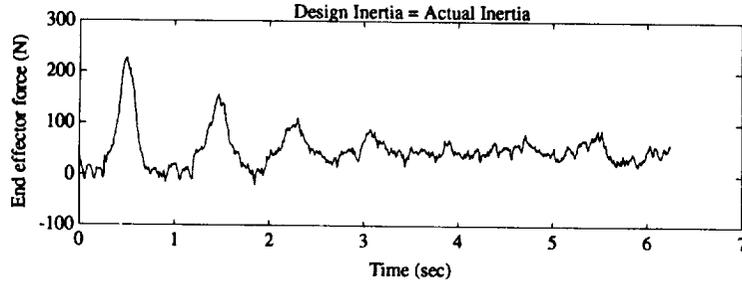


Figure 5: Impact with best estimate of inertia

using different inertias to compute the desired acceleration in the natural admittance controller. (The observer still used the best estimate of the inertia.) Values of one-half, one-quarter, twice, and four times the measured inertia were tried. Values for the endpoint stiffness and damping were also scaled to maintain a similar response. As expected, lower inertia values resulted in contact instability (figure 6). Contact using high inertia values was stable and tended to settle sooner (figure 7).

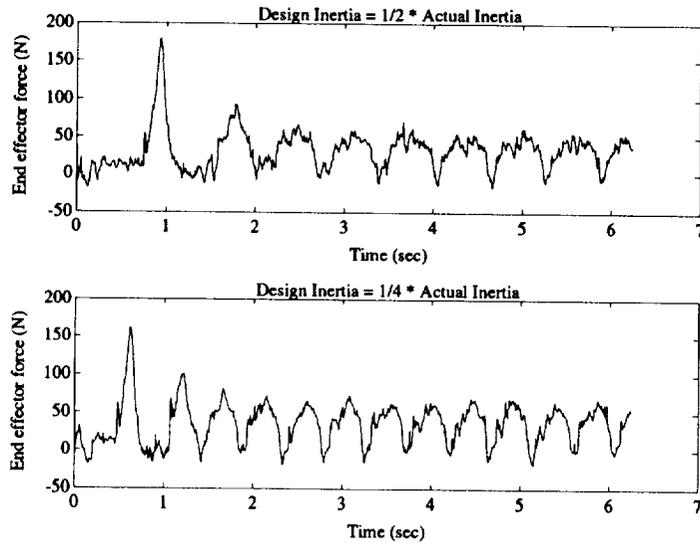


Figure 6: Impact with underestimated inertia

4 Discussion

Natural admittance control provides a mechanism for implementing force control in systems containing high levels of friction and transmission non-linearities in a manner that maintains passivity, provided that those undesirable characteristics are placed carefully in the mechanical design. Performance can approach the theoretical limit.

Our experiments indicate the beneficial effects of a final transmission stage that is low in inherent friction. It can be shown that there are fundamental restrictions on the

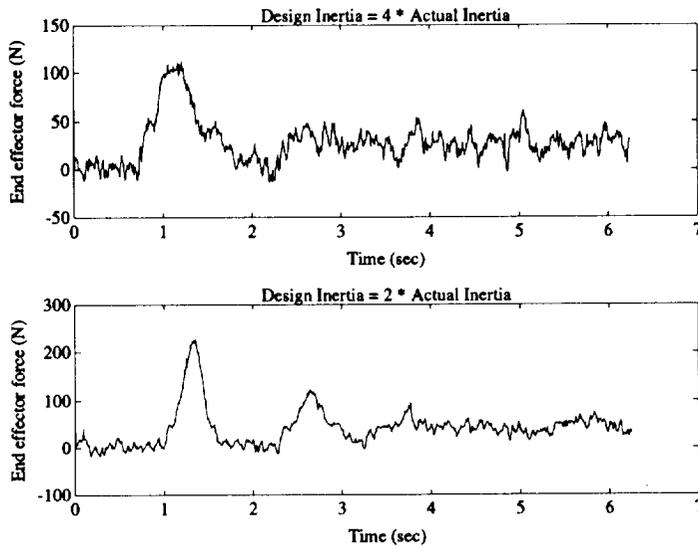


Figure 7: Impact with overestimated inertia

ability of a control algorithm to reject friction in a mechanical system. A particular problem occurs if the friction to be rejected is located far from the actuator coupled to it through a compliance, even a linear one. Trying to reject this “distal” friction can lead to an unstable system, showing the importance of designing mechanical systems with low inherent friction near the output, instead concentrating mechanical compliance (linear and non-linear), backlash, and non-linear friction such as Coulomb friction close to the actuator so that they can be successfully mitigated.

Also, as shown by this investigation, a crucial factor in the design of a controller is simply the availability of sensor signals. These are important for control model development, as well as for actual use by the control algorithms. Additionally, it is important to have at least one signal directly from the motor, either position or velocity.

5 Summary and Conclusions

A pitch-yaw joint was developed making use of a unique cone roller and Zerol bevel gear differential module. The use of rollers and gears in parallel combines the backlash-free operation of pure roller drives with the high torque capacity and lower bearing loads of gears. Reaction loads on the bearings of the roller-gear module were at least 2.5 times smaller than a similarly designed pure roller module. The variable preload devices required on pure roller drives were also replaced with a simpler spring-load system. The pitch-yaw module was driven by two gear motors via inline torque transducers. Rated load in either pitch or yaw was 186 N-m (1650 in-lbf). Mechanical efficiency of 97.5% was measured at 73% of rated output load, and zero backlash was found.

A force-control scheme modeled on natural admittance control was implemented on the drive system. The system exhibited good force control performance maintaining

stable interaction with a stiff environment. The natural inertia of the system was accepted and used as the desired inertia and friction was compensated through the natural admittance controller which made the manipulator track an ideal, frictionless manipulator. An observer greatly improved the performance of the velocity controller, allowing a four-fold increase in the loop gain. Interestingly, using the integrated velocity estimate for the position (integral velocity) feedback was also beneficial. The observer, in turn, was improved through the use of a friction estimator which included a load-dependent friction component and was fit to experimental data. As implemented, we successfully compensated for a large measure of the friction in the gear-motors in the presence of severe backlash.

The configuration of the joint module helped make the control more robust. The placement of the pitch-yaw module between the torque transducers and the environment limits the apparent environmental stiffness presented to the control by acting as a mechanical filter. However, any friction, backlash, or other undesirable mechanical effects located past the torque sensors could *not* be compensated for. Therefore the properties of the pitch-yaw module, with its low friction, smoothness, and low backlash, were crucial to the ultimate performance of the system.

Two mechanism design principles may be drawn from this for force-controlled manipulator systems. First, it is beneficial to have some form of mechanical filtering between the force or torque sensor and the environment. Second, since it is not possible to use the torque sensor to compensate for down-stream dynamics, it is important to make them as “friendly” as possible, i.e. put the most “ideal” (e.g.: low friction) components down stream of the sensor and put the components with undesirable dynamics (e.g.: high friction) up stream, closer to the actuator than the sensor is.

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